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Slipper Bearings and Vibration Control in Small Gas Turbines

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Several new principles of bearing design have proved effective in improving the reliability and smoothness of small gas turbines. The key development was the tilting-slipper thrust bearing which not only provided important radial clearance, but also made possible higher speeds at higher thrust loads. One of the unique features considered important for performance and low production costs is that the slippers float in their retainers instead of having fixed pivots.

Introduction and Résumé

The high rotor speeds of small gas turbines and other small turbine machinery have brought new bearing and vibration problems. Short and unpredictable bearing life, together with destructive high-frequency vibrations have been encountered, as rotor-tip speeds have been pushed upward to reduce size and weight, and to improve thermal efficiencies. Both difficulties have been found to be associated with rotor-bearing runout and rotor-balance errors, difficult, if not impossible, to eliminate with conventional balancing methods and conventional "antifriction" bearings.

This paper describes several new principles of bearing design which have proved effective in improving the reliability and smoothness of small gas turbines. The importance of radial clearance in all the rotor bearings, including the thrust bearing, has been established. A self-balancing action, or automatic compensation of balance errors has been obtained with increased running clearances in the bearings. The new bearings designed to accomplish this have virtually eliminated radial-bearing loads and the vibrations they cause.

The new bearings are modified forms of tilting-slipper radial-journal and axial-thrust bearings. The key development was the tilting-slipper thrust bearing which not only provided the required radial freedom, but also made possible higher speeds at higher thrust loads. One of the unique features considered important for performance and low production costs is that the slippers float in their retainers instead of having fixed pivots.

The bearings have been designed for use on conventional rotor shafts with case locations and mountings similar to ball and roller bearings. Because load capacities increase with speed with slipper bearings, journal diameters can be larger than with ball and roller types. The ability to use larger journals has helped make possible the higher compressor and turbine-tip speeds required for making the major advances in thermal efficiency needed is small gas turbines in order to reduce fuel consumption.

With proper attention to bearing design, method of lubrication, and oil flow, friction losses have been found to be less than with conventional sleeve bearings, and in fact, are in the same range as ball bearings. It is therefore not improper to refer to these slipper bearings as antifriction types.

About 5000 hours of testing in gas-turbine engines over the past five years, as well as many tests with special bearing test apparatus, have established these superior load-carrying capacities and antifriction characteristics. At the same time they are substantially free from critical production-tolerance problems of close fits, and tight trueness requirements often encountered with ball bearings for very high speeds.

The possibility of advantageous use of the same slipper-bearing principles in very small air-turbine machines, as well as in large gas turbines, has been explored in a preliminary way, with favorable conclusions. Photographs of typical examples and applications of the new bearings are shown in Figs. 1, 2, 3, and 4.

Fig. 1 First-stage rotor and bearings, 502-10C engine

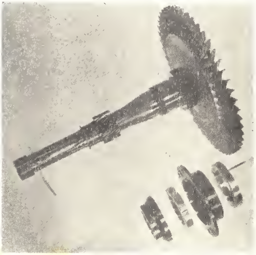

Fig. 2 Second-stage rotor and bearings, 502-10C engine

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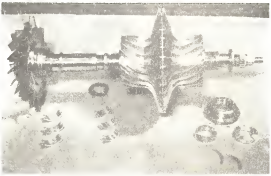


Fig. 3 Rotor and bearings, model 520 engine

(a) (b) (c) (d) (e)

Fig. 4 Exploded view of slipper thrust bearing, 502-10C engine (*a*, First-stage oil slinger, 6-91586-1; *s/n* 7; total time 332:27. *b*, First-stage radial bearing, 60-95068; *s/n* 8; total time 332:27. *c*, First-stage-compressor bearing retainer, 6-91589-1; *s/n* 7; total time 332:27. *d*, First-stage slipper bearing, 60-95067; *s/n* 8; total time 332:27. *e*, First-stage thrust sleeve, 60-95132; *s/n* 9; total time 332:27.)

Background

The theory, and application design principles, of plain journal and thrust bearings have, of course, been well established over a considerable period of years. So have the design criteria and characteristics of pivoted-shoe bearings originated in this country by Kingsbury and in England by A. G. M. Michell. Plain journal and thrust bearings have been very successfully used in the relatively high-speed mechanically driven superchargers of aircraft engines, also in exhaust driven turbochargers and the exhaust turbines of compound engines. Pivoted-shoe bearings have been widely used in large industrial-turbine machinery, and more recently in some of the large industrial gas turbines here and abroad.

But most of the smaller gas turbines and probably all of the small air-turbine machines used in jet airplanes have ball thrust bearings and ball or roller radial bearings. The reason for this preference on the smallest, and therefore the highest speed rotors, no doubt lies in the ready availability of the ball and roller bearings, and in the desire to avoid force-feed lubrication and scavenging systems; also in the widespread acceptance of the good antifriction characteristics of these bearings. In the light of the investigations described further on, this choice might be questioned. It would appear rather astonishing that these bearings have been made to operate with reasonable life at speeds of 50,000 rpm and above.

The original Boeing 502 gas-turbine engine of 1947 had a ball thrust bearing at the compressor end, and a roller bearing adjacent to the turbine. Because of a serious critical speed in the relatively flexible rotor shaft, a third bearing was added near the center of the shaft. Seizures of this center bearing occurred, and were eliminated by changing to the floating-sleeve bearing illustrated in Fig. 1. Later this same floating sleeve bearing was also substituted for the roller bearing at the turbine end. These two changes made a big improvement in reliabilty and life. The floating-sleeve bearings showed remarkable freedom from wear, and freedom from sensitivity to production tolerances. Furthermore, the heat-soak problem from quick engine shutdowns, which had seriously affected the previously used roller bearing near the turbine disk, was no longer present.

In preparing the advanced model of this engine, the 250-hp 502-10C which is now in small quantity production, there were two important factors requiring further improvement:

1 High-frequency vibration was causing failures of sheet-metal and other components of some engines in the field. These engines invariably showed vibration readings at first-stage rotor frequency of 20 *g*'s or higher. A production vibration tolerance of 15 *g*'s max had been established which alleviated these failures, but it was found extremely difficult and costly to achieve this smoothness in all engines. This was traced to slight changes in bearing journal "runout" of the shaft after the rotor assembly had been carefully balanced to less than 0.02 oz-in. balance error. No matter how carefully the rotor assembly was taken apart and reassembled in the engine, and despite all attempts to maintain super-accuracy of the individual pieces, it was not possible to maintain the journal runouts that existed during the balancing operation. Balance errors were therefore introduced in the engine at final assembly.

2 Development progress on the single-stage centrifugal compressor to higher efficiency, and higher pressure ratio for the new engine model, brought higher thrust loads and more frequent failures of the ball thrust bearing adjacent to the compressor. We were unable to get satisfactory assurance that a ball bearing could be made available to take this increased thrust load for even 500 hr. A serious look, beginning in 1954, was therefore directed toward the plain-type thrust bearing.

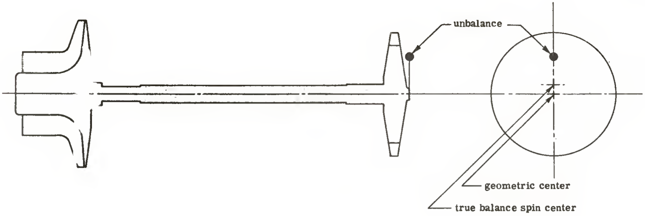
Sleeve Bearing Performance. At about this same time a very significant fact came to our attention and provided the clue to solution of both problems above mentioned. It was observed that loss of a turbine blade did not distress the floating-sleeve turbine bearings in spite of the substantial out of balance of the turbine wheel; nor was there evidence of excessive vibration. With the previous roller bearing, loss of a turbine blade invariably caused violent vibration of the engine and destruction of the bearing. It became clear that this superior behavior of the plain bearing was not, as we previously thought, due to higher load capacity but rather to the fact that the loads were small with the loose-fitting sleeve bearing, and very high with the close-clearance roller bearing. In other words, the turbine unbalance caused heavy bearing loads only when the bearing journal was closely restrained radially; and further, when this restraint on the bearing was removed, the engine vibration caused by the unbalance also disappeared. In the light of this discovery it seemed plausible that, in spite of the meticulous balancing operations carried out on the rotor assemblies, there were unbalance errors in the running engine that were translated into heavy bearing loads and vibration at rotor frequency at the thrust ball bearing.[2]

Self-Balancing With Loose Radial Clearance

The following principles became evident by analysis and were confirmed experimentally:

(*a*) *A rotating disk and shaft if not restrained by bearings will spin about its true balance center.* This simple "fact" gives some difficulty from a theoretical standpoint because current mathematical treatments of balancing, and critical speeds of shaft-

[2] Among the corrective measures investigated was electronic balancing of the rotor *after* engine assembly. This has been successful in mass-produced automobile engines, and should be possible in small turbines if a convenient production method could be worked out for adding or removing weight at the turbine disk without removing the turbine wheel.



Case 1 SPINNING ROTOR WITH NO BEARINGS

(assumed case of rotor spinning in space)

1. **Bearing Loads** - none
2. **Vibration** - none
3. **Critical Speeds** - none
4. **Effect of Unbalance** - spin center shifts from geometric center to true balance center

Fig. 5 Diagram, case 1, rotor with no bearings

mounted disks, require as a starting point that there be bearing supports. If we leave out the bearing supports we also leave undetermined the center of rotation, and therefore we cannot calculate either the unbalance or a critical speed. The author submits that this is prima-facie evidence that *if the bearings are loose enough there will be no unbalance nor will there be any critical speeds.* This is what our results indicate for turbine rotors, with either one or two disks. Fig. 5 illustrates this Case 1 of the rotor with no bearing restraints.

(*b*) *Relatively small amounts of unbalance will cause high bearing loads in small turbine machines if the bearing radial clearances are small.* This, of course, is because the centrifugal force of the unbalance increases as the square of the speed. For example, a 0.02 oz-in. unbalance in a turbine or compressor wheel running at 36,000 rpm will cause a bearing load of 46.2 lb if there is no clearance in the bearing

$$\text{Bearing load } P = \frac{W}{g} r\omega^2$$

$$= \frac{0.02}{32.2} \times \frac{1}{16 \times 12} \left(\frac{2\pi\ 36{,}000}{60}\right)^2$$

$$= 46.2 \text{ lb}$$

$$W = \text{lb unbalance at 1 in. radius}$$

$$g = 32.2 \text{ fps}$$

$$r = \text{radius 1 in.}$$

$$\omega = \frac{2\pi \text{ rpm}}{60} \text{ radians}$$

Note that 0.02 oz-in. unbalance is the present drawing limit for production balancing using the best available electronic balancing machines. If this amount of unbalance is exceeded by runout changes during assembly or engine operation, the bearing load and vibration will increase in direct proportion to the increase in unbalance. Loss of the tip of a turbine blade, for instance, wil produce an unbanace of about 1.0 oz-in., or 50 times the limit used on the drawing. The corresponding bearing load (if there is no clearance in the bearing) is 2310 lb, obviously beyond the capacity of the bearing, Fig. 6.

(*c*) *A small amount of bearing or shaft "runout" causes a relatively large amount of unbalance in a high-speed rotor.* Using the 0.02 oz-in. balancing limit of the above example, the amount of turbine-bearing eccentricity or journal runout to cause this amount of unbalance with a 13-lb turbine wheel would be equivalent to a center-of-rotation shift of 0.000096 in., or a runout of 0.000192 in. full indicator reading.

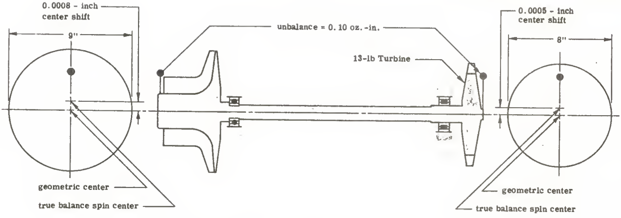
$$2C = \frac{W}{2(16 \times 13)} = \frac{0.02}{2(16 \times 13)} = 2\,(0.000096) = 0.000192 \text{ in.}$$

It has been found, however, that the runouts cannot be held closer than about 0.0005 in. (0.001 in. full indicator) as a production operation with conventional tolerances in the rotor pieces. Thus the unbalance error in the assembled engine is not the 0.02 oz-in. obtained on the balancing machine, but

$$0.02 \times \frac{0.001}{0.000192} \text{ or } 0.104 \text{ oz-in.}$$

The radial bearing load is then 231 lb, not 46.2 as mentioned above. This is the beginning situation without any allowance for distortions in the running engine from centrifugal stresses or temperature gradients, Fig. 6.

(*d*) *The amount of radial shift to allow self-balancing for relatively large unbalance errors, is small.* This follows from (*c*). With the 8-in-diam, 13-lb turbine disk running at 36,000 rpm, of the previous example, the center of rotation need shift only 0.005 in. to regain true balance from a 1 oz-in. unbalance.



Case 2 CONVENTIONAL ROTOR WITH BALL AND ROLLER BEARINGS

(8-lb compressor and 13-lb turbine spinning at 36,000 rpm)

1. Bearing Loads - compressor bearing radial load = 231 lb
turbine-bearing radial load = 231 lb

2. Vibration - compressor end - 29 g's at 600 cps frequency
turbine end - 18 g's at 600 cps frequency

3. Critical Speed - severe fundamental critical at 26,000 rpm

4. Effect of Unbalance - destructive engine vibration and short bearing life

Note that with the rotor perfectly balanced an eccentricity of only 0.0005 inch at the turbine bearing and 0.0008 inch at the compressor bearing will cause this same unbalance of 0.10 ounce-inch and bearing load of 231 lb.

Fig. 6 Diagram, case 2, rotor with ball and roller bearings

W = unbalance when rotated on geometric center = 1 oz-in.
w = rotor weight = 13 lb
C = radial shift of true balance center away from geometric center to put rotor back in balance

$$W = 16 \times w \times C$$

$$1 \text{ oz-in.} = 16 \times 13 \times C$$

$$C = \frac{1}{16 \times 13} = 0.005 \text{ in.}$$

Thus the 0.007 total radial clearance used in the floating-sleeve bearing is in the range to accommodate 0.005-in. shift of the rotor running center to allow self-balancing for a 1-oz-in. balance error, equivalent to loss of half a turbine blade, Fig. 7.[3]

(*e*) *Provision for radial clearance to obtain self-balancing is especially significant in a thrust bearing.* This is because a ball bearing when used to take thrust will automatically take up all the ball clearance manufactured into the bearing. Fig. 8 illustrates this. Thus a ball thrust bearing is inherently a zero-clearance bearing, and therefore any balance errors or runout errors are reflected immediately in higher bearing loads. For example, with 0.0002 production runout tolerance for the inner ball race, the bearing itself will create a radial load of 30 lb, using an 8-lb compressor-impeller disk at 36,000 rpm. With the rotor balanced to 0.02 oz-in., we then have an additional radial bearing load of 46.2 lb or a total of 76.2 lb. If the total shaft runout goes to the production limit of 0.001 in. full indicator when the engine is assembled, the bearing load then becomes 226 lb. This built-in bearing load will produce vibration of about 20 g's at 600 cycles frequency. And this is without any allowance for distortions due to temperature, or shaft runout errors due to running shift of the bolted-up parts. The design thrust load, in this case 370 lb, when combined with the 226-lb radial load, makes a difficult situation for a ball bearing running at 36,000 rpm.

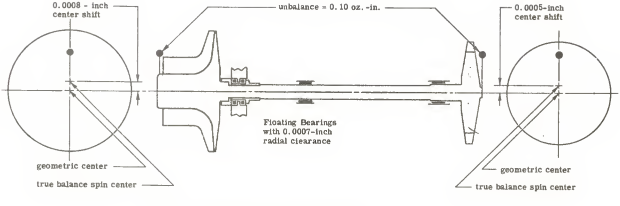
From these considerations it becomes apparent that thrust bearings for high-speed shafts should have radial freedom to avoid the "built-in" overload. The slipper thrust bearing of Fig. 4 does this by allowing the slipper to float with the same radial clearances and double oil film as the floating-sleeve radial bearings previously mentioned.[4]

(*f*) *The amount of self-balancing obtained with the floating radial and thrust bearings already mentioned is enough to eliminate the need for balancing the assembled rotor.* In other words, if each individual component of the rotor assembly is dynamically balanced—which ordinarily should be the case—the rotor can then be assembled in the engine without having to go through the meticulous and expensive assembly-balancing operation.

The great value of this feature in reducing engine-production costs will be obvious. Rotor parts can be stocked as details instead of assemblies, and "match marking" of pieces can be eliminated. Even more important is the simplification of main-

[3] This suggests that increased ball and roller clearances will increase the safety factors for these bearings by allowing self-balancing for small balance errors. This undoubtedly is true where no thrust load is present. But large clearances of the order under discussion are usually distasteful to antifriction bearing manufacturers because the load capacity is reduced and ball retainer difficulties may occur.

[4] It is, of course, possible to mount a ball thrust bearing so that there is radial clearance between the outer race of the bearing and its engine case retainer. This, however, involves motion against the thrust lip which then becomes a plain thrust-bearing problem. It would therefore seem proper to handle the entire thrust bearing as a plain-type thrust bearing. Soft materials such as synthetic rubber have been used between the ball bearing outer race and its retainer and bearing loads from unbalance errors can undoubtedly be reduced in this manner, but not eliminated.



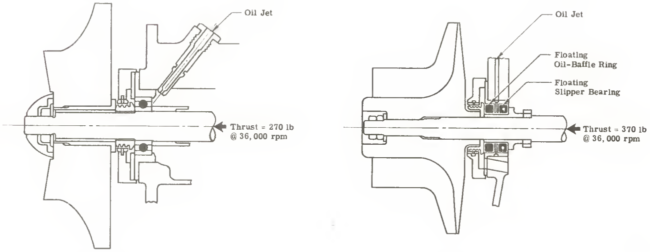
Case 3 ROTOR WITH FLOATING BEARINGS

(8-lb compressor and 13-lb turbine spinning at 36,000 rpm)

1. Bearing Loads - none
2. Vibration - negligible
3. Critical Speeds - none
4. Effect of Unbalance - no effect until unbalance exceeds 1.0 oz.-in. or 10 times the 0.10 oz.-in. of the example, i.e., the 0.007-in. bearing clearance will accommodate the center shift of 0.005 in. required by 1.0 oz.-in. unbalance.

Fig. 7 Diagram, case 3, rotor with floating bearings

Model 502-8 (1954) | Model 502-10C (1957)



BALL THRUST BEARING

Zero radial clearance in bearing when balls are taking thrust load because of angular contact between balls and races.

SLIPPER THRUST BEARING

0.006 to 0.011 inch radial clearance when slippers are taking thrust load.

Fig. 8 Assembly drawing cross sections, ball and slipper thrust bearings

tenance operations in the field because spare compressor wheels and turbine wheels can be stocked and installed without going back to the factory or special base where an electronic balancing machine is available. Experience with assembly of about two hundred Boeing 502-10C engines with floating rotor bearings has already established the feasibility of this interchangeability of rotor components. None of these rotors were assembly-balanced and the maximum rotor frequency-vibration readings have been in the 1 to 5-g range—well below the production tolerance requirement of 15 g's.

Fig. 9 compares the vibration of one of these engines, first with a ball thrust bearing, second with a floating-slipper thrust bearing, and third with a known unbalance of 2.2 grams (0.078 oz) introduced by installing a headed screw at the impeller hub. The

Fig. 9 [illegible] thrust bearings

same rotor [illegible] com-pressor [illegible] the shaft [illegible] engine. It will [illegible] from a ball bearing [illegible] rotor fre-quency [illegible] an unbalance [illegible] pro-duction [illegible]

Design and [illegible]

The fiv [illegible] all rotor bea [illegible] the thrust bea [illegible] established [illegible] bearings.

A rather [illegible] out to dete [illegible] arrange-ment. [illegible] were radial fre [illegible] load capacity, [illegible] cost in [illegible] en-gine prices.

Fig. 10 [illegible] bearing designs tested [illegible] several plain thrust [illegible] sur-faces to pre [illegible]

Fig. 10 Bearing test machine

Fig. 11 Split-ring plain thrust and radial bearing

Fig. 12 Fixed-[illegible] thrust bearing

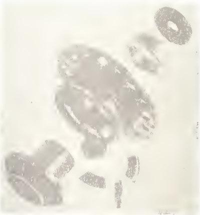


Fig. 13 Experimental hydrostatic-type slipper thrust bearing

ing-slipper thrust bearing with fixed pivots, and a separate floating-sleeve radial bearing. Fig. 13 is one of several hydrodynamic types tested. The floating-slipper-type radial bearings selected for use in the 520 engine model, shown in Fig. 3, have differential arcs inside and outside to permit rocking. In another new model a fixed tilting-slipper radial bearing has been used successfully.

Floating-Slipper Thrust Bearing

The thrust-bearing design shown in Fig. 4 was selected for production use because it showed superior load capacity and low friction. It also is the simplest and the easiest to manufacture. Reliability has been good, and it has the ruggedness and relative insensitivity to production tolerances common to plain-type

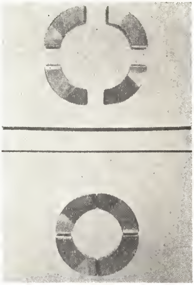


Fig. 14 Close-up view of floating-slipper segments and assembly

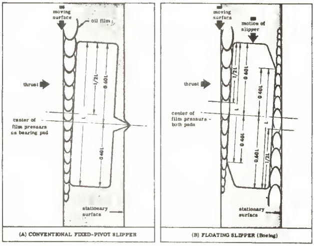


Fig. 15 Diagram, graphic representation of operation of fixed and floating slippers

journal bearings. Details of the slipper assembly are shown in Fig. 14.

One serious fault of most pressure-fed plain bearings, particularly in the case of high-speed bearings as required for turbomachinery is the variation of oil flow with slight variations in clearance due to production tolerances or wear. This defect is not present in the new floating-slipper bearings because the oil flow is jet controlled, not clearance controlled. Once the jet sizes are established, there is almost complete freedom from oil-flow control problems.

This bearing design was not worked out overnight. A great deal of test work on the bearing machine and in the engine was required, with a number of errors and many failures, before the critical design criteria were uncovered. One of the big difficulties in this work was the fact that when a bearing failed at 36,000 rpm or similar high speed, destruction of the parts of the bearing was so rapid, usually from the heat generated, that the exact cause of the failure was often impossible to determine by examination. Thermocouples were useful in determining operating-temperature levels, but proved of little value in anticipating failures or preventing destruction of the bearing surfaces.

The important design criteria were found to be the following:

1 As anticipated, the characteristics of the tilting-pad thrust bearing were found to be ideal for gas turbines and other turbomachinery where thrust increases more or less directly with speed. Referring to the left-hand sketch in Fig. 15, the load capacity of a fixed-pivot tilting-pad bearing has been established by Michell.[5]

$$P = \frac{0.0669\ Uu\ A}{C^2}$$

where P = load in psi
U = speed
u = viscosity
A = pad area
c = inclination of pad to runner plane

Also: Position of center of pressure (pivoting point) = 42 per cent of pad length from trailing edge. That is, the pivoting point and center of pressure on the pad should be at a location 42 per cent of the length of the pad from the trailing edge.

These equations show that the load capacity varies directly with speed and oil viscosity, and inversely as the square of the inclination. Also that "the efficiency and effectiveness of the bearing are therefore enhanced by making the angle of inclination as small as possible." Further, the location of the pivot point does not vary appreciably with the width, length, or inclination of the bearing pad (that is, the 42 per cent point from trailing edge is correct for all shapes of pad and all angles of tilt).

2 From these considerations (item 1) it is evident that a floating slipper pad as represented by the right-hand sketch of Fig. 15 will operate just as well as the fixed-pivot type, except that the load capacity and the friction loss will be less because of the reduced relative speed between the slipper and the shaft runner on one side, and between the slipper and stationary thrust collar on the other side.

In practice, the correctness of this concept has been confirmed except that the following additional design criteria have been established:

(*a*) Because of the arcuate shape of the thrust pads it is essential that the trailing edges of the pads on opposite sides of each bearing segment be cut parallel to each other, and symmetrical to the segment. These are the "hinge points" for tilt, and if they are not parallel the thrust will cause cocking of one or both pads relative to their respective running surfaces. All attempts to get good load capacity with floating-thrust slippers were unsatisfactory until this important point was established.

(*b*) The amount of offset of the thrust pads on either side of each segment can be greater than indicated in Fig. 15 with satisfactory results, but where relatively heavy loading occurs at the lower speeds, the 40-per-cent rule provides higher load capacity with less friction.

(*c*) Centrifugal loading of the floating segments on the outer radial bearing surfaces, due to their own weight, becomes important at high speeds. Because of this, and to help promote axial tilt for the thrust pads without binding at the ends, the inner and outer circumferential surfaces are made with "differential" radii similar to the radially loaded slippers previously mentioned. This has the further advantage of permitting the thrust slippers to take also radial loads.

(*d*) While the thrust-load carrying capacity of the floating slipper is theoretically less than that of a fixed slipper because of the reduced relative speed under the same operating conditions, in practice this is of no consequence because the diameter of the

[5] "Lubrication," by A. G. M. Michell, Blackie and Son, London, England.

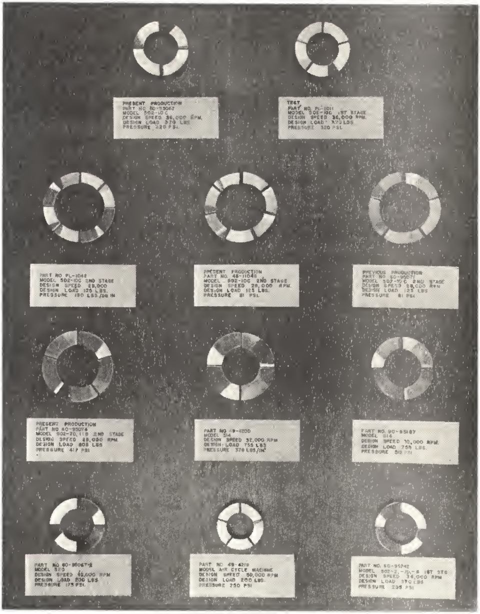


Fig. 16 Display of various sizes and applications of floating-slipper bearings

bearing and pad areas can readily be increased to handle the required load. The floating-slipper bearing is being used successfully with pad pressures as high as 800 psi. Floating speed of these bearings has been checked as about $^1/_3$ of the shaft speed.

(*e*) For easier handling, and to retain the bearing segments in matched widths, a garter spring retainer is used, threaded through drilled holes in the segments.

Fig. 16 shows a display of various sizes and applications of this bearing. Included is one with alternate left and right hand-cut segments, which can take thrust in either direction of rotation. By using two such bearings, one on either side of the thrust collar, thrust can be taken in either direction as well. Furthermore, radial loads can be taken by both bearings, and it is impossible to improperly assemble the parts. The latter can be an important consideration because it is not obvious to most people by visual inspection of the "one-way" bearings of Figs. 4 and 14, which directions of rotation and thrust apply, nor is a simple method of marking apparent.

Fig. 17 shows test curves made on the bearing machine, comparing the friction torque losses of the floating-slipper bearing and a precision ball bearing under the same thrust loadings, the speed being constant at 36,000 rpm. These curves show the importance of keeping oil flow as small as possible consistent with safe operating temperatures. The coefficients of friction corresponding to these curves are in the range of 0.002 to 0.006 for both the slipper and ball bearings. It would appear appropriate, therefore, to apply the term antifriction bearing equally well to the slipper-type bearings as to the ball and roller bearings.

Application of the Floating Bearing Principle to Very Small and Very Large Rotors

Having shown the advantages of floating bearings for a small gas turbine of the 250-hp class, the question arises as to whether the same would be true for smaller or larger rotors.

To investigate this from the standpoint of applicability of the self-balancing principle for balance errors, three rotor sizes were

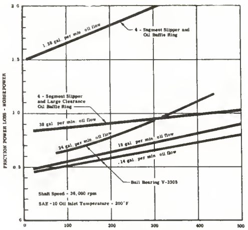


Fig. 17 Curve plot, friction losses of slipper and ball thrust bearings compared by bearing machine tests

assumed, one having the 8-in. diam, 13-lb turbine of the example already described, one having a 4-in. turbine, or $^1/_2$ the size, and one having a 24-in. turbine or 3 times the size. Assuming the tip speeds are the same, and that the amount of unbalance error to be compensated for is a constant percentage of the rotor weight, it was readily shown that the required amount of center shift of the true balance center from the geometric center, is identical in all three cases. In other woids, the amount of center shift to obtain self-balancing, does not change with rotor-scale size as long as the amount of unbalance is a fixed proportion of the rotor weight. This is true whether the rotor weight in the examples be varied as the cube of the diameter or as the square of the diameter. Thus the amount of bearing float or diametral clearance required to get self-compensation for the same percentage of unbalance error, is constant regardless of scale size (that is, 0.007-in. bearing clearance of our 250-hp-engine example will give the same results for any engine size).

Following similar procedure for bearing loads it will be found that the bearing total loading from the unbalance error is also identical for all sizes of rotor, assuming no radial clearance. Since bearing sizes will vary approximately as the scale size, it becomes evident that *bearing unit pressures will increase directly as the decrease in scale size,* for the same percentage of unbalance error. In other words, where ball thrust bearings are used, with the resulting lack of any radial freedom, the problems of overloading and vibration due to unbalance errors get increasingly more critical as the size decreases.

We reach the general conclusions, then, that floating thrust bearings will be just as effective in very large and very small machines, in eliminating vibration and bearing problems, as in the 250-hp machine described above, but the need is most critical in the smaller machines.

Among other questions that have been brought up relative to the self-balancing action of floating bearings, the effect of relatively large bearing clearances on seal and blade-tip clearances is worth mention. Blade tip clearances in the engines using the slipper bearings were already relatively large (0.025 in.) to avoid tip rubs from shroud-ring distortion, and no difficulties have occurred from the 0.006 to 0.010 bearing clearances. The labyrinth seals originally caused some difficulty until materials were used that permitted rubbing without seizure or melting. It has been found that seal rubs can readily be detected by excessive vibration readings, no doubt because the seal when it rubs is acting as a zero clearance bearing.

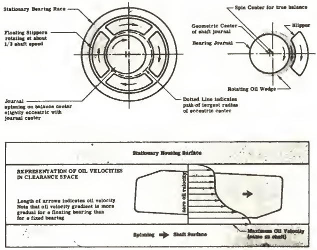


Fig. 18 Diagram, explanation of self-balancing action of floating bearings, by oil-viscosity behavior

Another question rather difficult to answer with positive assurance is: "How does a bearing journal run eccentrically in a floating bearing without causing hydraulic loading or oil whip?" Fig. 18 is offered to explain this in terms of oil velocities in the clearance space. It is suggested that the oil film between the journal and the floating bearing does not, in fact, have to change its thickness 36,000 times a minute, as might be assumed, but rather, in effect the oil film follows the shaft rotation.